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PRELIMINARY EVALUATION OF GREASES TO 316° C AND SOLID LUBRICANTS TO 816° C IN BALL BEARINGS

by Harold E. Sliney and Robert L. Johnson

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ABSTRACT

A special apparatus designed for the evaluation of high-temperature lubricants in 20-mm-bore ball bearings is described. ~~The results of bearing runs at~~ temperatures up to 1500° F (816° C) in air are reported. At 450° and 600° F (232° and 316° C), a fluorocarbon grease lubricated the 440-C steel bearings for longer times than did polyphenyl ether and silicone greases thickened with dyes and/or MoS_2. Cobalt alloy ball bearings, lubricated either with barium fluoride - calcium fluoride coatings bonded to the cages or with porous metal cages impregnated with these fluorides, ran successfully at 1200° and 1500° F (650° and 816° C) under a thrust load of 30 lb (134 N) and at a shaft speed of 5000 rpm.

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SUMMARY

Ball bearings fabricated from high-temperature alloys (Stellite 6B balls and races and René 41 or composite cages) were operated in air at temperatures up to 1500° F (816° C). The adherent oxides which formed on these alloys prevented bearing seizure and high wear on otherwise unlubricated bearings. However, bearing torque was high and erratic. The use of calcium fluoride - barium fluoride solid lubricant formulations provided the additional lubrication required for low torque and stable bearing operation. to insert

The bearings were lubricated with either 0.002-inch- (0.05-mm-) thick fluoride coatings bonded to the bearing cages or by means of self-lubricating composite bearing cages. The composite material consisted of a porous nickel alloy matrix with a fluoride lubricant filler.

The bearings were 204 size (20-mm bore) of the angular contact type with a 20° contact angle and 0.002 inch (0.05 mm) internal clearance. Under a 30-pound (134-N) thrust load, fluoride-lubricated bearings ran at 5000 rpm without failing for as long as 700 hours at 1200° F (650° C) and 149 hours at 1500° F (816° C). The shortest lives under these conditions were of the order of 20 to 50 hours. Wear rates were very low until just prior to the onset of bearing failure.

In the case of bearings with fluoride-coated cages, the failures which occurred were initiated by a small accumulation of loose wear particles in the ball-raceway contact region. This resulted in rough operation, increased torque, and a self-accelerating wear process. Therefore, it is important to provide a means of solid particle rejection in the design or in the method of installation of solid-lubricated ball bearings.

There was little tendency for particulate matter to accumulate in bearings with composite cages. Instead, failures were caused by a gradual cage distortion which eventually closed the cage-race clearances.

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INTRODUCTION

The attainment of adequate wear life and acceptable reliability are two of the most difficult problems in the area of solid lubrication. When a solid lubricant is used as a bonded coating, its wear life is finite; when the coating wears out, the lubricant cannot be readily replaced. Self-lubricating composites often extend wear life capabilities but are susceptible to distortion and cracking. Therefore, after basic studies have demonstrated that a solid has promising friction and wear characteristics, it is then of interest to determine the performance of the candidate lubricant in actual bearings.

In this report, a new type of bearing test apparatus, which was designed for evaluating solid lubricants or greases in 204 size (20-mm bore) ball bearings, is described. The preliminary test results for greases at bearing temperatures from 325° to 600° F (163° to 316° C) and for solid lubricants at bearing temperatures from 1200° to 1500° F (650° to 816° C) are presented.

The solid lubricants were applied as bonded coatings on the bearing cages, or, in some cases, the cages were fabricated from self-lubricating composites consisting of porous, sintered nickel-chromium alloy (Inconel) impregnated with a calcium fluoride - barium fluoride (CaF_2-BaF_2) eutectic filler. All tests were conducted in air. A thrust load of 30 pounds (134 N) and shaft speeds of 2000 or 5000 rpm were used. The calculated maximum Hertz stress at 1200° F (650° C) for a 30-pound (134-N) thrust load was 93 000 psi (6.4×10^8 N/m^2).

BACKGROUND

Ball bearings lubricated with solid lubricants offer several important advantages: (1) they can be operated at temperature extremes beyond the capabilities of oil- or grease-lubricated bearings (refs. 1 and 2); (2) because dry-lubricated bearings do not require cooling, recirculating oil systems with their associated pumps and heat exchangers are not needed; and (3) rotating shafts can be shortened because dry-lubricated bearings can be located closer to heat sources.

These advantages are of importance in general machine design, but they are of particular importance in aerospace equipment which must be of light weight and minimum complexity.

Several techniques for the use of solid lubricants in ball bearings have been studied. One method that has shown promise is powder lubrication, whereby the solid film is continuously deposited on the bearing surfaces from a suspension of solid particles in a carrier gas. Some early studies, which proved the feasibility of lubricating rolling contact bearings in this manner employed molybdenum disulfide (MoS_2) or graphite with air

as the carrier gas (refs. 3 and 4). Molybdenum disulfide provided lubrication to about 700° F (370° C) before decomposition of the lubricant occurred; graphite provided lubrication to 1000° F (538° C). At this temperature, graphite oxidized to harmless gaseous products but was replenished by the continuous supply of lubricant. Other experiments showed that MoS_2 could be effectively used to at least 1000° F (538° C) if an inert carrier gas (nitrogen) were employed (ref. 5).

It has been shown that the variation in the friction coefficient of graphite over a large temperature range can be reduced by the addition of certain metal oxides; cadmium oxide is a particularly beneficial additive (ref. 6). Cadmium oxide - graphite powder in an air carrier has lubricated cobalt-alloy ball bearings at temperatures from room temperature to 1000° F (538° C) (ref. 7). Very careful metering is required to provide adequate lubrication while avoiding lubricant starvation or excessive lubricant buildup in the bearings.

Ball bearings have been effectively lubricated in air at temperatures up to 750° F (400° C) with MoS_2 bonded to the bearing cage (ref. 8). In reference 8 it is also shown that the wear life can be substantially improved by providing a number of depressions or lubricant reservoirs in the ball pockets and the locating lands of the bearing cages.

Lead oxide - lead silicate (PbO-$4PbO \cdot SiO_2$) solid lubricant coatings have lubricated ball bearings at 1000° and 1200° F (538° and 650° C) (ref. 9). The melting point of the PbO-$4PbO \cdot SiO_2$ coatings limits their maximum useful temperature to about 1250° F (677° C).

Another class of solid lubricant coatings for severe environments is the series of fused fluorides reported in reference 10. The CaF_2-BaF_2 coatings, which are evaluated as lubricants for bearing cages in this study, have much higher melting points than PbO coatings and have been shown to lubricate at temperatures up to 1500° F (816° C). These coatings also have somewhat better friction and wear-life characteristics than PbO coatings at temperatures below 1000° F (538° C) but have considerably higher friction coefficients than MoS_2 below about 700° F (370° C).

Solid lubricants may also be incorporated into a composite bearing material. Some examples are MoS_2 with silver (ref. 11), molybdenum diselenide ($MoSe_2$) with gallium and indium (ref. 2), $MoSe_2$ with polytetrafluoroethylene (PTFE) and silver (ref. 12), and MoS_2 with a metal matrix of iron or platinum (ref. 13).

The composites described in reference 12 were evaluated as cage materials for ball bearings. The bearing performed satisfactorily in vacuum over a temperature range of -180° to 300° F (-118° to 149° C) (ref. 14). Tungsten diselenide (WSe_2) and gallium composite cages (ref. 2) provided lubrication in air to 950° F (510° C). The composites of porous nickel chromium alloy (Inconel) impregnated with CaF_2-BaF_2 eutectic, which is evaluated as a self-lubricating bearing cage material in this study, have shown prom-

ise in fundamental sliding-friction and wear studies for uses at temperatures to 1500° F (816° C) (ref. 15).

TEST BEARINGS

The solid-lubricated bearings used were 204 size, angular-contact ball bearings. The ball set consists of twelve 9/32-inch- (7.15-mm-) diameter balls. Both inner and outer raceway curvatures are 54 percent, and the nominal contact angle is 20°. The internal clearance (radial play) is 0.002 inch (0.05 mm). Ball and race material is a cobalt-chromium alloy (Stellite 6B) hardened to Rockwell C-42. The cage is inner-race-located. Some of the cages are a nickel-chromium alloy (René 41) coated with 0.001- to 0.002-inch (0.03- to 0.05-mm) thickness of fluoride solid lubricant (60 wt. % CaF_2 - 40 wt. % BaF_2); others are self-lubricating composites of porous, sintered nickel-chromium alloy (Inconel) impregnated with eutectic fluoride (38 wt. % CaF_2 - 62 wt. % BaF_2). Cage-ball and cage-race diametral clearances are variable from 0.010 to 0.030 inch (0.25 to 0.75 mm). In each bearing, the cage clearances are held to a tolerance of 0.0005 inch (0.013 mm).

The grease lubricated bearings are deep-groove ball bearings of hardened 440-C steel. The cages are stamped steel, and the bearings are provided with two press-fit steel grease shields.

BEARING TEST APPARATUS

A cutaway view of the bearing test head is given in figure 1. The test head can be completely assembled as a unit prior to mounting on the drive system; it consists of a double concentric bellows assembly provided with a bearing housing at the front and back for the test bearing and the slave bearing, respectively. The annular space between the two bellows is gas-tight, and pneumatic pressure is used to apply a uniform thrust load to the outer races of the two bearings. As indicated in the figure, the inner races are located on the shaft by means of a tubular spacer between the two bearings. The bellows were calibrated on the fixture shown schematically in figure 2. The calibration curve is given in figure 3.

Pressurized gas is supplied to the bellows through a long section of small-diameter, 1/16-inch (1.6-mm) tubing which does not significantly interfere with torque measurements.

The assembled test head is mounted on a motor-driven support bearing spindle. Alinement and shaft balance are achieved during fabrication by performing final machin-

ing and balancing of the shaft while mounted on the same spindle. During a bearing test, the combined torques of the test bearing and the slave bearing apply a rotational moment to the outer races and hence to the bearing housing. Rotation is prevented by a swiveled linkage between a torque arm on the housing and a strain ring which continuously measures the combined torque. A variable-speed drive provides a range of shaft speeds from 500 to 5000 rpm.

Heating is provided by a high-frequency (475 000 Hz) induction coil. The test-bearing housing and a hub which extends into the bore of the inner race are heated directly and conduct heat into the bearing races. Bearing outer-race temperatures are measured with Chromel-Alumel thermocouples. Both outer and inner race temperatures are monitored with an infrared pyrometer capable of measuring temperatures from 250° to 4500° F (120° to 2500° C).

The bearing test head was carefully designed to minimize heat transfer from the heated test bearing to the slave bearing by keeping the cross sectional area for conductive heat transfer to a minimum.

A cross section through the bearing test head and its temperature profile are given in figure 4. In addition to thermocouples and the infrared pyrometer, temperatures were determined with temperature indicating marking sticks. These merely bracketed temperature within 100° F (55° C) increments, but they were useful at the cooler end of the test head where the surfaces were unoxidized and highly reflective and, therefore, of uncertain infrared emissivity. At a test-bearing outer-race temperature of 1200° F (650° C), the rear-bearing outer-race temperature was about 400° F (200° C). Figure 5 gives the temperature of the slave bearing for test-bearing temperatures up to 1300° F (700° C).

RESULTS AND DISCUSSION

Grease-Lubricated Bearings

Several greases, which are useful to 600° F (316° C) have been reported in reference 16; therefore, greases were considered for lubrication of the slave bearing. In our studies, long-duration runs were made with a number of high-temperature greases to gain some idea of the life and torque characteristics of these greases in the 440-C steel bearings of the type used as slave bearings in the test section.

The greases evaluated were

(1) A 5P4E polyphenyl ether fluid thickened with an organic dye

(2) A polyphenyl ether (unknown molecular weight) - silicone blend fluid thickened with an organic dye and MoS_2

(3) A perfluorinated alkyl ether fluid thickened with an organic dye

(4) A perfluorinated alkyl ether fluid thickened with a fluorocarbon telomer (intermediate molecular weight polymer)

Long-Duration Tests of Grease-Packed Bearings

The failure criterion for the long duration tests was arbitrarily set at a bearing torque of 10 inch-ounces (7.0×10^{-2} N-m). This corresponds to about five times the typical combined torque of the two bearings during normal operation.

The results of the long-duration tests are summarized in table I. Both polyphenyl ether greases tended to harden during prolonged operation at bearing temperatures from 325° to 600° F (163° to 316° C).

The hardening of the grease was accompanied by an increase in bearing torque. The bearings did not fail from loss of lubricating material, but rather because of stiffening of the grease. No bearing damage was noted, but the high torque and rough operation after less than 50 hours at bearing temperatures of 385°, 450°, and 600° F (196°, 232°, and 316° C) made these bearing-grease combinations unsatisfactory for purposes of this program.

The two fluorocarbon greases performed in a more satisfactory manner. Lives in excess of 200 hours were obtained at bearing temperatures up to 450° F (230° C) with both the dye-thickened and the telomer-thickened grease. At high temperatures, life was limited by evaporation of the lubricating fluid.

At 500° F (260° C), the telomer-thickened fluorocarbon grease lubricated for 33 hours. Inspection of the bearing showed that most of the fluid had evaporated or bled out of the bearing. The residue was a thin white film of what appeared to be the telomer thickener on all of the internal surfaces of the bearing. No bearing damage was apparent except for a small amount of wear on several of the cage ball pockets where the protective film had worn through. At 600° F (316° C), the dye-thickened grease lubricated for 20 hours. Inspection of the bearing indicated that the fluid had evaporated. After the fluid had evaporated, lubrication was apparently provided by the dye thickener, which remained in the bearing.

The telomer-thickened fluorocarbon grease was chosen as the slave-bearing lubricant. It was chosen in favor of the polyphenyl ether greases because of longer bearing life and more stable torque characteristics. It was chosen in favor of the dye-thickened fluorocarbon grease only because it is clean and convenient to handle.

Grease Torque Characteristics

In these experiments, both the test and the slave bearings were packed with 3 cubic centimeters of telomer-thickened fluorocarbon grease. Figure 6 gives the test-bearing torque at 2000 and 5000 rpm, and at outer-race temperatures up to 500° F (260° C). After starting a test and with no external heat addition, the bearing temperature slowly increased to a constant stable value. Bearing torque was continuously monitored during this time. The bearing temperature stabilized at 160° F (71° C) at 2000 rpm, and at 210° F (99° C) at 5000 rpm. The bearing torque stabilized at 2 inch-ounces (1.4×10^{-2} N-m) at 2000 rpm, and at 3.5 inch-ounces (2.5×10^{-2} N-m) at 5000 rpm. During these measurements, the temperatures of the test and the support bearings were equal within 20° F (11° C). Therefore, the torque per bearing was assumed to be one-half the combined torque of the two bearings.

After stabilization temperatures and torque were determined with no external heat addition, the bearings were stopped and allowed to cool to room temperature. They were then restarted, and the test bearing was rapidly induction heated to a stable, elevated temperature. The slave bearing, which was heated only by frictional heat and by conduction from the test bearing, reached a stable temperature at a much lower rate than the test bearing. The combined torque was measured while the slave-bearing temperature was still within the range for which the torque was previously determined. The torque of the heated test bearing was then obtained by difference. By repeating this process at several test-bearing temperatures, the torque of the grease-lubricated bearing was determined over the temperature range shown in figure 6. This figure was used in subsequent tests as a calibration curve to correct test-bearing torque for the torque contribution of the slave bearing.

Solid-Lubricated Bearings

Coatings. - The ball bearings in this series of tests were lubricated with a 0.002-inch- (0.05-mm-) thick coating of 60 weight percent CaF_2 plus 40 weight percent BaF_2 bonded to the bearing cages. The tests were run at 1200° and 1500° F (650° and 816° C). The results are summarized in table II. Test-bearing torque was in the range of 1 to 4 inch-ounces (7.0×10^{-3} to 28.0×10^{-3} N-m) and was, therefore, comparable to the torque of the grease-lubricated bearings. The bearings were considered failed when the torque increased to 10 inch-ounces (7.0×10^{-2} N-m), which is the same failure criterion that was chosen for the grease-lubricated bearings.

The potential for long life is indicated by the results of experiment 1 in table II, in which a bearing heated to 1200° F (650° C) ran for 220 hours at 2000 rpm and an addi-

tional 710 hours at 5000 rpm and did not fail. The earliest failure at 1200° F (650° C) for a bearing equipped with an inner-race-located cage occurred after 56 hours at 2000 rpm (experiment 2).

Bearing life at 1500° F (816° C) (experiment 5) was shorter than at 1200° F (650° C), but the bearing performed well for 12 hours at 2000 rpm and an additional 24 hours at 5000 rpm. The balls and races were in good condition with a highly polished appearance on the rolling contact surfaces. However, the solid lubricant coating on the cage had become rough and porous. It was reported in reference 10 that the coatings were relatively unaffected by long-duration exposure to air at 1200° F (650° C), but at 1500° F (816° C), the coating slowly deteriorated because of oxidation of the base metal at the coating bond line. The coating is not adversely affected by exposure to 1500° F (816° C) in nonoxidizing environments such as hydrogen or argon.

Examination of the bearings after the tests indicated that all failures were associated with the generation of wear debris from the balls and the raceways. The coatings in the ball pockets were apparently abraded away, but the coatings on the race-located surfaces of the cages were highly polished and apparently in excellent condition.

The bearings ran smoothly as long as particulate matter did not become trapped in the ball-race contact regions. However, if a small accumulation of solid particles collected in the raceway, bearing operation became rough, torque increased, and the wear process became self-accelerating. The source of the initial particles was possibly excess coating material which had been worn off the ball pockets. The coating material functions as a lubricant when bonded to the cage or when transferred as an adherent film to contacting surfaces. However, loose particles of the same material, when jammed into the ball-raceway contacts, could easily disturb the polished surface film and cause removal of surface oxides. The loose oxide particles then act as true abrasives. X-ray examination of the loose wear debris found in failed bearings indicated that it was primarily cobalt oxide.

The use of thinner lubricant coatings than the 0.002-inch- (0.05-mm-) thick coatings used in this study may be beneficial in minimizing the formation of loose lubricant particles. It has been observed that thick films rapidly wear down to a film thickness of about 0.0002 inch (0.005 mm), particularly for Hertzian contact configurations (ref. 15). The remaining thin film may then be abraded by particles jammed into the contact regions. Loose lubricant particles do not have this effect on the performance of solid lubricants in basic lubrication studies, which employ a pin-on-disk specimen configuration, because loose particles are immediately swept out of the contact region.

The experiments reported herein were preliminary in nature and insufficient in number to give a reliable indication of the bearing life that can be reasonably expected. However, they demonstrated the feasibility of lubricating cobalt-chromium alloy ball bearings

with fluoride solid lubricant coatings at 1200° and 1500° F (650° and 816° C) for useful periods of time.

Further, these experiments defined an important problem area - the need to minimize the formation of loose particles in the bearing and to provide a means of removing from the contact regions any particles which do form.

Self-lubricating composite cages. - Ball bearings equipped with self-lubricating composite cages were also tested. The results are summarized in table III. In nine experiments at bearing temperatures from 1200° to 1500° F (650° to 816° C), only one bearing failed from lack of lubrication. This was during experiment 6 (table III), where the composite cage was used in the as-machined condition. During machining, the metal phase of the composite material was smeared over the lubricant phase. Because of the lack of lubricant at the surface, the bearing immediately ran at a torque in excess of 10 inch-ounces (7.0×10^{-2} N-m).

Several surface treatments were effective in reexposing the lubricant after machining; these included acid etches, wet honing with waterproof sandpaper, and heat treatment to promote fluoride exudation to the etched or honed surface. The heat treatment was performed in a nitrogen or argon atmosphere. In general, better results were obtained in 1200° F (650° C) bearing tests with cages that had been heat treated in argon. The lowest wear of all bearing elements was obtained when the etched cage was coated with a thin burnished overlay of CaF_2-BaF_2 (experiment 5, table III).

In experiment 1(table III), the bearing ran at 1200° F (650° C) and 5000 rpm for 149 hours with no failure and with very low wear. Another bearing (experiment 9) ran at 1500° F (816° C) and 2000 rpm for 20 hours and at 5000 rpm for an additional 50 hours. The other bearings ran for shorter periods of time, with the shortest duration at 23 hours. Failure of these bearings was not initiated by loose wear particles. In all cases except experiment 6, failure was caused by closing of cage clearances due to dimensional instability of the composite material. The dimensional instability was a gradual uniform swell of the composite during exposure to air at high temperatures, and does not appear to be associated with the mechanical stresses on the cage during operation of the bearing. Because failures were caused by cage distortion and not by wear or lack of lubrication, very long endurance life should be attainable if the problem of dimensional instability can be solved.

Figures 7 and 8 give the torque-time characteristics under various conditions for bearings equipped with self-lubricating cages. The torque of an unlubricated bearing with a machined nickel-chromium alloy (René 41) cage is shown for comparison.

Very high and erratic bearing torque was characteristic of the unlubricated bearing. However, examination of the bearing after this test showed that in spite of the unfavorable torque, the bearing was still in good condition. The bearing was run under essentially a pure thrust load and was carefully alined. Under these conditions, the forces

acting on the cage at the cage-ball and cage-race sliding contact areas are small (ref. 17), and the naturally formed oxide films on Stellite 6B and René 41 are apparently adequate to prevent surface damage (wear). In air, at elevated temperatures, therefore, the primary purpose of the fluoride lubricant is to provide a low and relatively steady bearing torque.

CONCLUDING REMARKS

The new type of bearing test apparatus described in this report was satisfactory for evaluating high-temperature greases or solid lubricants in thrust-loaded ball bearings.

Ball bearings with Stellite 6B balls and races were lubricated at 1200° and 1500° F (650° and 816° C) with calcium fluoride - barium fluoride solid lubricants. These lubricants were effective either as bonded coatings on René 41 alloy bearing cages or as fillers in composite bearing cages.

Perfluorinated alkyl ether greases with telomer or organic dye thickeners were satisfactory lubricants up to 500° F (260° C) for the 440-C tool-steel slave bearings used in the test rig.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, April 19, 1968,
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TABLE I. - RESULTS OF TESTS WITH BALL BEARINGS

LUBRICATED BY HIGH-TEMPERATURE GREASES

[Bearing type, deep-groove, eight-ball, stamped-cage; bearing material, 440-C steel; amount of lubricant in bearing, 3 cm^3; double press-fit grease shield; atmosphere, air; thrust load, 30 lb (134 N); failure criterion, increase of bearing torque to 10 in.-oz (7.0×10^{-2} N-m).]

Grease description		Bearing race temperature		Shaft speed, rpm	Test duration, hr (a)	Typical torque (b)	
Fluid	Thickener	^{o}F	^{o}C			in.-oz	N-m
5P4E Polyphenyl ether	Organic dye	325	163	2000	220	2.0	14×10^{-4}
		385	196	5000	46	1.5	11
		450	230	5000	45	1.0	7
Polyphenyl ether-silicone blend	Organic dye and MoS_2	325	163	5000	148	1.2	8×10^{-4}
		385	196	5000	23	1.0	7
		600	316	5000	3	1.8	13
Fluorocarbon oil	Organic dye	450	230	5000	384	1.0	7×10^{-4}
		600	316	5000	20	3.0	21
Fluorocarbon oil	Fluorocarbon telomer	325	163	2000	[c]134	1.2	8×10^{-4}
		400	204	2000	215	1.0	7
		325	163	5000	[c]734	1.4	10
		450	230	5000	209	1.0	7
		500	260	5000	33	.8	6

[a]Lubricant failed in times indicated unless otherwise noted.

[b]Stable torque before onset of failure.

[c]No failure in times indicated.

TABLE II. - PERFORMANCE OF HIGH-TEMPERATURE BALL BEARINGS

EQUIPPED WITH FLUORIDE-COATED CAGES

[Bearing size, 204 (20-mm bore); contact angle, 20°; number of balls, 12; bearing material, Stellite 6B (age-hardened cobalt-chromium alloy); cage material, René 41 (age-hardened nickel-chromium alloy) coated with approximately 0.002 in. (0.051 mm) CaF_2-BaF_2 solid lubricant; thrust load, 30 lb (134 N); atmosphere, air; failure criterion, increase of bearing torque to 10 in.-oz (7.0×10^{-2} N-m).]

Experiment	Cage clearance				Temperature		Shaft speed, rpm	Test duration, hr (a)	Typical torque		Weight-change rate, mg/hr		
	Inner race		Ball pocket		°F	°C			in.-oz	N-m	Inner race	Outer race	Balls
	in.	mm	in.	mm									
1	0.010	0.25	0.016	0.41	1200	650	2000	220	2	14×10^{-3}	------	-----	------
	------	-----	-----	----	----	---	5000	[b]710	3	21	-0.01	-0.02	-0.06
2	0.010	0.25	0.016	0.41	1200	650	2000	56	1	7×10^{-3}	-18.0	-7.6	-6.2
3	0.025	0.64	0.020	0.50	1200	650	2000	140	1	7×10^{-3}	-0.06	-0.04	[c]-0.04
4	[d]0.013	[d]0.33	0.016	0.41	1200	650	588	5	$1\frac{1}{2}$	11×10^{-3}	------	-----	------
	------	-----	-----	----	----	---	1500	20	2	14	------	-----	------
	------	-----	-----	----	----	---	2000	2	4	28	------	-----	------
	------	-----	-----	----	----	---	5000	6	4	28	------	-0.26	-0.10
5	0.010	0.25	0.016	0.41	1500	816	2000	12	3	21×10^{-3}	0.9	-0.16	0.16
	------	-----	-----	----	----	---	5000	24	4	28	-2.0	-1.0	-3.2

[a]Failed in time indicated unless otherwise noted.

[b]No failure in time indicated.

[c]Hastelloy-C (nickel-chromium alloy) balls.

[d]Outer race-cage clearance.

TABLE III. - PERFORMANCE OF HIGH-TEMPERATURE BALL BEARINGS EQUIPPED WITH SELF-LUBRICATING COMPOSITE CAGES

[Bearing size, 204 (20-mm bore); contact angle, 20°; number of balls, 12; bearing material, Stellite 6B (age-hardened cobalt-chromium alloy); cage material, sintered Inconel (nickel-chromium alloy) of 35 vol. % porosity vacuum-impregnated with CaF_2-BaF_2 eutectic; thrust load, 30 lb (134 N); atmosphere, air; failure criterion, increase of bearing torque to 10 in.-oz (7.0×10^{-2} N-m).]

Experiment	Cage pretreatment					Cage clearance				Temperature		Shaft speed, rpm	Test duration, hr (a)	Typical torque		Weight change rate, mg/hr		
	Surface treatment	Heat treatment				Inner race		Ball pockets		$^{\circ}F$	$^{\circ}C$			in.-oz	N-m	Inner race	Outer race	Balls
		Atmosphere	$^{\circ}F$	$^{\circ}C$	hr	in.	mm	in.	mm									
1	Acid etch	Argon	1500	816	2	0.020	0.50	0.016	0.41	1200	650	5000	[b]149	2.6	18×10^{-3}	-0.04	-0.03	-0.02
2	Acid etch	Argon	1500	816	2	0.015	0.38	0.018	0.46	1200	650	2000	41	1.0	7×10^{-3}	-----	0.02	0
	----------	--------	----	---	---	-----	----	-----	----	----	---	5000	15	5.0	35	-0.03	-.05	0
3	Acid etch	Nitrogen	1600	870	2	0.015	0.38	0.016	0.41	1200	650	2000	60	4.5	31×10^{-3}	-0.03	-0.03	-0.08
4	Acid etch	Nitrogen	1600	870	2	0.010	0.25	0.016	0.41	1200	650	5000	23	4	28×10^{-3}	-0.17	-0.03	-----
5	Acid etch, CaF_2-BaF_2 overlay	Nitrogen	1700	930	0.3	0.027	0.69	0.015	0.38	1200	650	2000	27	1.5	11×10^{-3}	0	0	0
6	None	None	----	---	---	0.020	0.50	0.016	0.41	1200	650	2000	0	10	70×10^{-3}	-----	-----	-----
7	Acid etch and hone	None	----	---	---	0.030	0.75	0.020	0.50	1300	700	2000	34	1	7×10^{-3}	-2.1	-0.9	-0.01
8	Acid etch	Nitrogen	1600	870	2	0.020	0.50	0.016	0.41	1400	760	2000	35	1	7×10^{-3}	-0.01	0.06	0
9	Acid etch	Argon	1500	816	2	0.020	0.50	0.016	0.41	1500	816	2000	[b]20	0.6	4×10^{-3}	0.13	0.16	0.09
	----------	--------	----	---	---	-----	----	-----	----	----	---	5000	[b]50	1.4	9.8	-.17	-.04	-.32

[a]Failed in time indicated unless otherwise noted.

[b]No failure in time indicated.

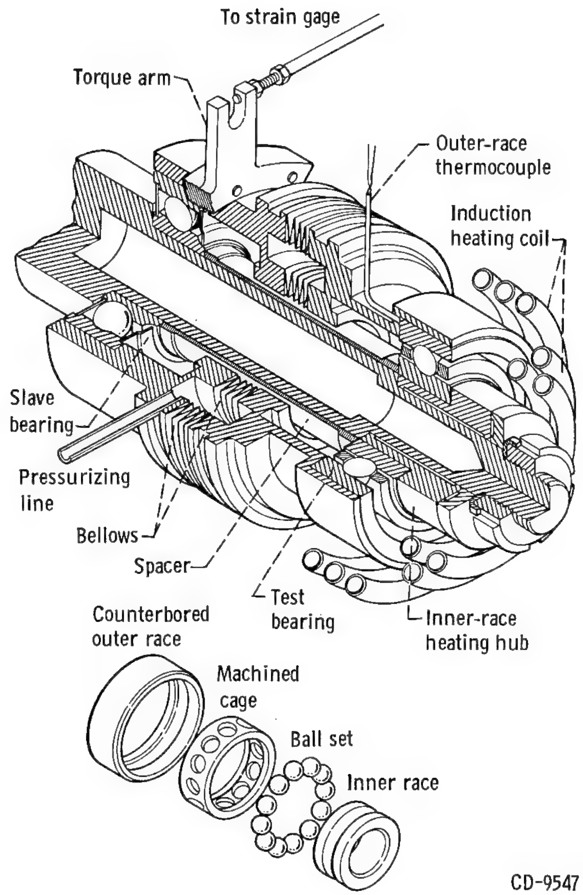

Figure 1. - Bearing test head and exploded view of test bearing.

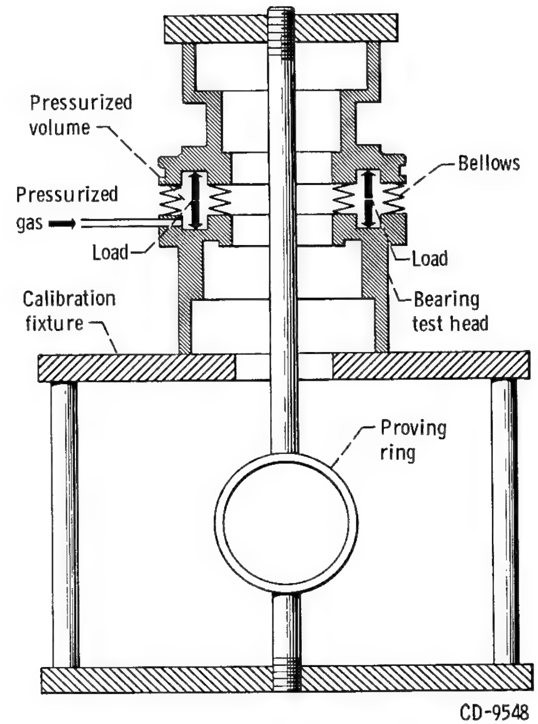

Figure 2. - Bellows calibration fixture.

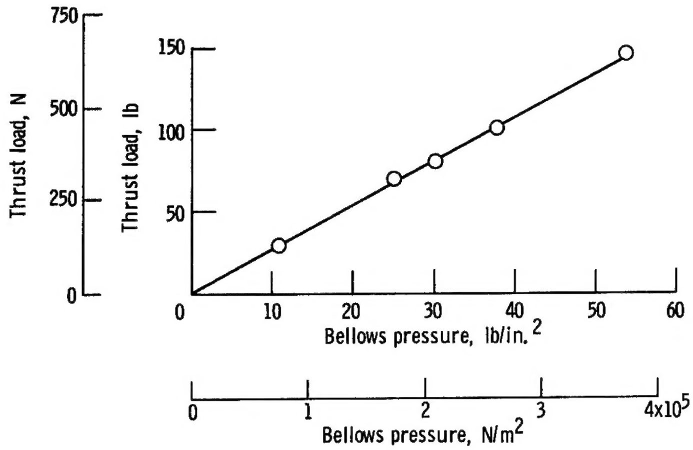

Figure 3. - Calibration curve for bellows thrust-loading device.

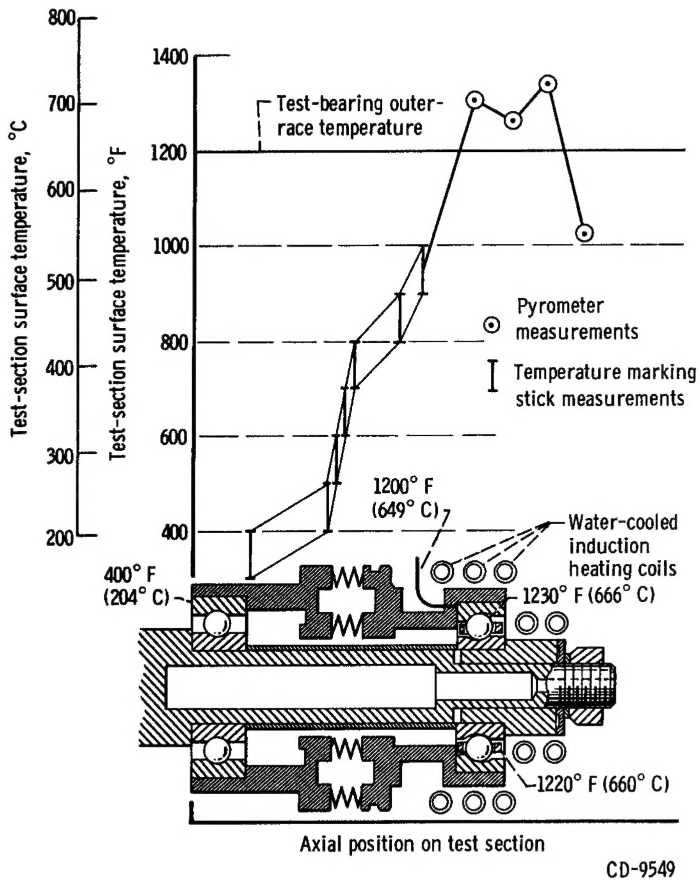

Figure 4. - Temperature distribution of bearing test section.

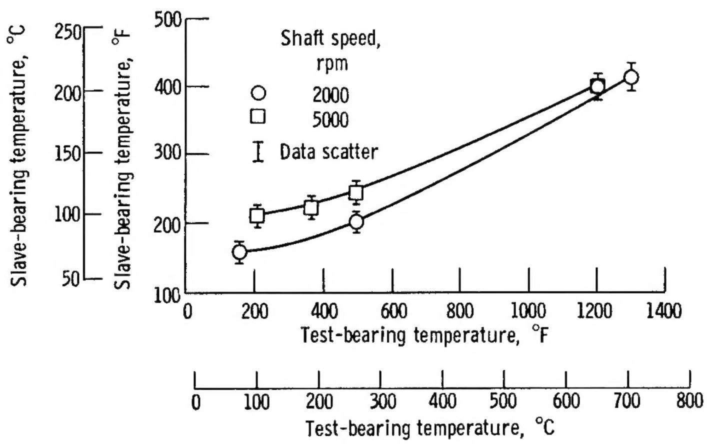

Figure 5. - Relation of slave-bearing temperature to test-bearing temperature. Thrust load, 30 pounds (134 N).

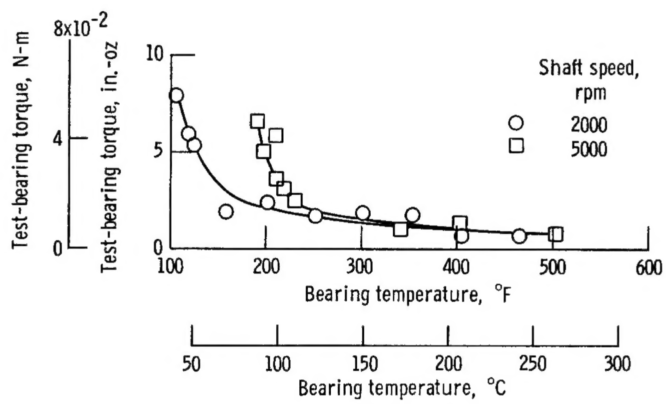

Figure 6. - Torque of 20-millimeter ball bearings lubricated with a fluorocarbon grease. Thrust load, 30 pounds (134 N).

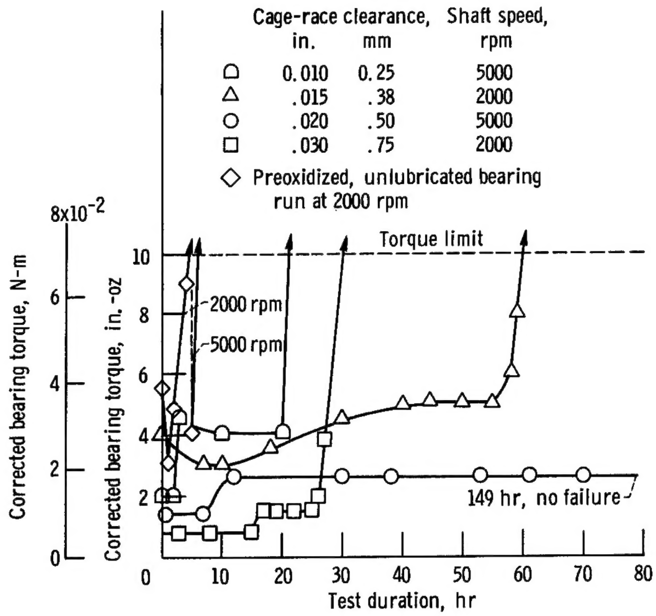

Figure 7. - Performance of cobalt alloy ball bearings with self-lubricated fluoride-Inconel composite cages. Inner-race-located cage; thrust load, 30 pounds (134 N); ball pocket clearance, 0.016 inch (0.41 mm).

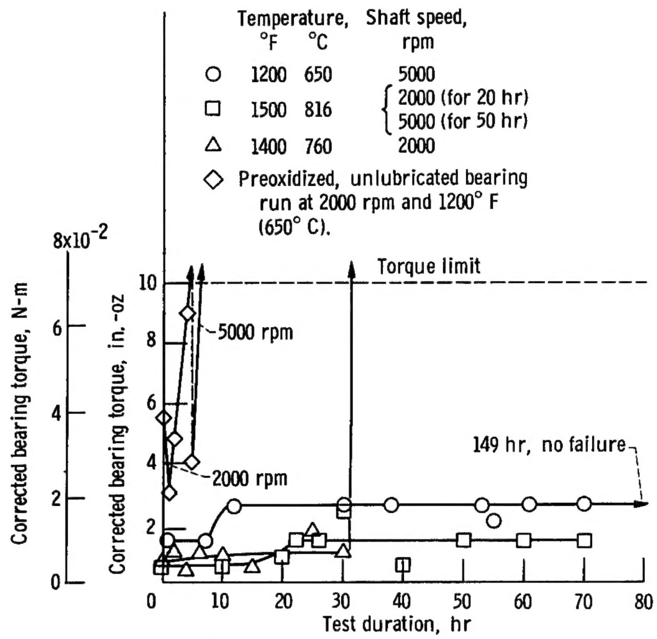

Figure 8. - Influence of bearing temperature and speed on performance of cobalt alloy ball bearings with self-lubricated fluoride-Inconel composite cages. Inner-race-located cage; cage-race clearance, 0.020 inch (0.50 mm); thrust load, 30 pounds (134 N); ball pocket clearance, 0.016 inch (0.41 mm).

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